Design and Manufacturing of Carbon Fiber Composite Drive Shaft as an Alternative to Conventional Steel Drive Shaft

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Abstract:- Weight is a critical factor in performance of all Automobiles, especially in the racing sector. Drive shafts are one of the heavier components in automobiles and a reduction in their weight can increase vehicle performance considerably. Replacing conventional drive shafts with drive shafts made of composite material is a feasible way of performance enhancement. Optimization of composite material used, leads to further reduction in weight thus leading to better vehicle performance. Our paper deals with the optimization of carbon fiber composite drive shaft of SAE Baja vehicle for the best possible weight reduction without compromising its strength. The major factors in design of composite shaft include the number of layers of Carbon fiber and the orientation of fibers within each layer. Using the MATLAB software and finite element analysis, optimization was carried out for all the possible layers. After the design finalization, the shaft has been manufactured and tested. The data obtained by testing of composite shaft was compared with that of the steel drive shaft. The result was in coherence with the aim of this project.

Keywords:- *Drivetrain; Transmission; Driveshaft; Carbon Fiber; Composite; Torque; Epoxy; Laminas; Resin; Spool; ABD Matrix; Laminator; Filament Winding; Ply-by-ply.*

I. INTRODUCTION

A. Background

The function of automotive drive shaft is to transmit the torque that is produced from the engine to the rear wheels to push the vehicle forward and reverse. The drive shaft is used to transfer this torque from differential to wheels that cannot be connected directly because of distance or the need to allow for relative movement between them. The steel drive shaft is usually manufactured in two pieces to increase the fundamental bending natural frequency because the bending natural frequency of a shaft is inversely proportional to the square of beam length and proportional to the square root of specific module. As torque carriers, drive shafts are subjected to torsion and shear stress, equivalent to the difference between the input torque and the load, they must therefore be strong enough to bear the stress, whilst avoiding too much additional weight as that would in turn increase their inertia. Since carbon fiber epoxy composite materials have more than four times specific stiffness of steel or aluminium materials, it is possible to manufacture composite drive shafts in lesser weight without compromising the strength.

B. Composite materials

A composite material (also called a composition material or shortened to composite) is a material made from two or more constituent materials with significantly different physical or chemical properties that, when combined, produce a material with characteristics different from the individual components.

C. Fibers

Fibers are the principal constituent in a fiber-reinforced composite material. They occupy the largest volume fraction in a composite laminate and share the major portion of the load acting on a composite structure. Proper selection of the type, amount and orientation of fibers is very important, because it influences the following characteristics of a composite laminate.

II. LITERATURE REVIEW

The first application of composite drive shafts to automobiles was developed by Spicer U—Joint divisions of the Dana Corporation for the Ford Econoline van models in l985. Dr Andrew Pollard GKN Technology, Wolverhampton, UK has been delivering carbon composite propeller shafts to production car applications since 1988. The technical drivers for use of composite material in this application are very powerful; the weight saving in particular can be considerable. Despite this, the overwhelming majority of automotive propeller shaft continue to be produced from steel.

III. SPECIFICATION OF THE PROBLEM

The drive shaft diameter should not exceed 50mm due to space constraint. The drive shaft of transmission system is to be designed optimally for following specified design requirements which are as follows. The following specifications are assumed which are based on literature and available standards of automobile drive shaft.

BAJA CAR Specifications:
Model: BAJA ATV (2016 Season)
Max Engine Output: 10 HP @ 3800rpm
Max Torque: 18.85 Nm @2600rpm

In this project based the student built BAJA ATV for SAE Baja Competitions, the inner diameter of the drive shaft is taken as 30mm and length of the shaft is 400mm.

IV. DESIGN OF STEEL DRIVESHAFT

A. *Mechanical Properties*

The three design specifications like torque transmission capability, buckling torque capability and bending natural frequency should be satisfied by the steel drive shaft. Steel is used for automotive drive shaft applications. The material properties of the AISI4130 are given in table.

Mechanical Properties	Symbol	Unit	Values
Young's Modulus	E	GPa	210
Shear Modulus	G	GPa	80
Poisson's Ratio	μ	-	0.3
Density	ρ	Kg/m^3	7800
Yield Strength	S_y	MPa	660

Table 1:- Mechanical Properties of steel (AISI4130)

B. *Gear Ratio for BAJA ATV*

Parameters	Values
Primary Reduction	1
CVT Low ratio	3.83
CVT High Ratio	0.9
Final Reduction	7.8306

Table 2:- BAJA Car Gear ratios

So, the torque is maximum at CVT lower ratio range, when the speed of vehicle is low. Therefore the maximum torque will be,

$$Tmax = 59 \times 3.83 \times 7.8306 = 565.334Nm$$

This will be available at the end of open differential. So on each shaft torque will be half of this calculated torque value due to use of open differential.

$$Mt = \frac{Tmax}{2} = \frac{565.334}{2} = 282.667Nm$$

C. *Design of steel shaft based on Torsional Strength basis*

Torsional Strength: The primary load in the drive shaft is torsion. The maximum shear stress τ_{max} in the driveshaft is at outer radius, and is given as follows considering fos = 2.5;

$$\tau = \frac{16 \times 2.5 \times Mt}{\pi \times d_0^3 \times (1 - C^4)}$$

$$\frac{0.5 \times 600 \times 10^6}{2} = \frac{16 \times 2.5 \times 282.667}{\pi \times 37^3 \times (1 - C^4)}$$

C = 0.8105

Which gives, D_i = 0.8105 x 37 = 29.989 mm ≈ 30 mm

$$Thickness\ t = \frac{(D_o - D_i)}{2} = \frac{(37 - 25)}{2} = 3.5mm$$

$$Mass\ of\ Steel\ drive\ shaft = \rho \times \left(\frac{\pi}{4}\right) \times (Do^2 - D_i^2) \times L$$

$$= 7800 \times \left(\frac{\pi}{4}\right) \times (37^2 - 30^2) \times 0.4$$

$$= 1.15K$$

D. *Design based on Torsional Rigidity*

$$\theta = \frac{584 \times M_t \times L}{G \times D_0^4 \times (1 - C^4)}$$

$$\theta = \frac{584 \times 282.667 \times 0.4}{(80 \times 10^9) \times 0.037^4 \times (1 - 0.8105^4)}$$

$$\theta = 2.3243^0$$

The permissible angle of twist for machine tool application is 0.25 degree per meter length. For line shaft in between 4 degree to 5 degree in the limiting value.

E. *Torsional Buckling Capacity of Drive Shaft*

$$T_b = (2\,\pi\, r_m{}^2\, t)(0.272 \times E)\left(\frac{t}{r_m}\right)^{\frac{3}{2}}$$

$$T_b = (2\,\pi\,(0.016747^2) \times 0.0035)(0.272 \times 210 \times 10^9)\left(\frac{3.5}{16.747}\right)^{\frac{3}{2}}$$

$$T_b = 33784.149\ Nm$$

The value of Critical torsional buckling moment is larger than the applied torque of 282.667Nm. Thus the shaft will withstand the torsional buckling capacity such that the condition is satisfied.

F. *Natural Frequency Calculations*

Natural frequencies can be found by using two theories:

1. Timoshenko Beam Theory
2. Bernoulli Euler Theory

Out of these two theories, as per the assumptions Bernoulli Euler Theory is more suitable in this case.

Bernoulli Euler Theory:

It neglects both transverse shear deformation as well as rotary inertia effects. Natural frequency based on the Bernoulli Euler Theory is given by,

$$f_{nt} = \frac{\pi \times P^2}{2 \times L^2}\sqrt{E \times \frac{I_x}{m_l}}$$

Where,

f_{nt} = Natural frequency based on Bernoulli Euler theory, Hz

P = 1, First Natural Frequency

I_x = Area moment of inertia in X direction, m^4

Ml = Mass per unit length, Kg/m

Now the area moment of inertia, I_x is

$$I_x = \left(\frac{\pi}{4}\right) \times (r_0^4 - r_i^4)$$

$$I_x = \left(\frac{\pi}{4}\right) \times (0.0185^4 - 0.015^4)$$

$$I_x = 5.22934 \times 10^{-8}\ m^4$$

Mass per unit length of the shaft is,

$$m_l = \pi \times (r_0^2 - r_i^2) \times \rho$$

$$m_l = \pi \times (0.0185^2 - 0.015^2) \times 7800$$

$$m_l = 2.877\ Kg/m$$

To get the natural frequency following Bernoulli Euler Formula is,

$$f_{nt} = \frac{\pi \times 1^2}{2 \times 0.4^2}\sqrt{210 \times 10^9 \times \frac{5.22934 \times 10^{-8}}{2.877}}$$

$$f_{nt} = 606.538\ Hz$$

This value is greater than the minimum desired natural frequency of 60 Hz. Thus, the steel design of solid shaft of diameter 24.2035mm is an acceptable design and is comparable with OEM shaft dimension of 23mm.

The critical speed of shaft is given by,

$$N_{crt} = 60 \times f_{nt}$$

$$N_{crt} = 60 \times 606.538$$

$$N_{crt} = 36392.3\ rpm.$$

G. Analytical Results

Sr. No.	Parameters	Values
1	Outer Diameter	37 mm
2	Inner Diameter	30 mm
3	Thickness	2.5 mm
4	Applied Torque	282.667 Nm
5	Natural Frequency	606.583 Hz
6	Torsional Buckling Capacity	33788.149 Nm
7	Critical Speed	36352.3 rpm
8	Mass	1.15 Kg

Table 3:- Analytical Results of Steel Drive Shaft

V. DESIGN OF COMPOSITE DRIVESHAFT

A. Mechanical Properties of Carbon Fiber

Carbon Fiber (ZOLTEK PANEX 35) reinforced composites are remarkable in their performance characteristics and properties that include: high strength, low weight, high stiffness, corrosion resistance, heat resistance, and electrical conductivity.

Material Properties	Unit	Values
Number of Filament	-	12,000
Tensile Strength	MPa	4,278
Tensile Modulus	GPa	240
Elongation	%	18
Density	g/cm^3	1.8
Filament Diameter	-	7 µ

Table 4:- Mechanical Properties of Carbon Fiber (PANEX 35)

B. Mechanical Properties of Epoxy Resin (Araldite LY 1564/Hardener XB 3486)

Epoxy resin LY1564 is a modified low viscosity bisphenol-A based liquid epoxy resin for filament winding. It is highly flexible. The reactivity may easily be adjusted to demands through the combination of the hardeners. The hardener XB 3486 is used as it has a long pot life which helps in the production of very large industrial parts.

Mechanical Properties	Unit	Values
Aspect	Visual	Clear colorless to slightly yellow liquid
Density at 25°C	g/cm^2	1.1-1.2
Viscosity at 25°C	mPa-s	1200-1400
Epoxide Eq. Weight [EEW]	gm/eq	161-173
Tensile Strength	Mpa	72-76
Tensile elongation at break	%	4.6-5.0
Tensile modulus	Mpa	2860-3000
Flexural Strength	Mpa	118-130
Flexural elongation at break	%	5.5-6.5
Flexural modulus	Gpa	2900-3050
Shear Strength	Mpa	53-58

Table 5:- Mechanical properties Epoxy resin(LY-1564/Aradur 3486)

C. Micromechanical Analysis of Lamina

- Volume Fraction of Fiber: 0.7 (70%)
- Volume fraction of Matrix: 0.3 (30%)
- Volume of composites: 1 (100%)

Density of Composite:

$$\rho_c = \rho_f \times V_f + \rho_m \times V_m$$

From the above table, we know

$$\rho_f = 1800\ Kg/m\hat{}3\ ,\ \rho_f = 960\ Kg/m\hat{}3$$

$$\rho_c = 1800 \times 0.7 + 960 \times 0.3 = 1548\ Kg/m\hat{}3$$

Weight of Fiber:

$$W_f = \frac{\rho_f}{\rho_c} \times V_f$$

$$W_f = \frac{1800}{1548} \times 0.7$$

$$W_f = 0.8139$$

Weight of Matrix:

$$W_m = \frac{\rho_m}{\rho_c} \times V_m$$

$$W_m = \frac{960}{1548} \times 0.3$$

$$W_m = 0.1860$$

Sum of mass Fractions:

$$w_f + w_m = 0.8139 + 0.1860$$

$$w_f + w_m = 0.99994$$

$$w_f + w_m = 1$$

Young's Modulus of Lamina:

$$E_1 = E_f \times V_f + E_m \times V_m$$

From properties table

$$E_f = 240\ GPa$$

$$E_m = 3\ GPa$$

$$E_1 = 240 \times 0.7 + 3 \times 0.3$$

$$E_1 = 168.9\ GPa$$

Young's Modulus of Lamina (Transverse Direction):

$$\frac{1}{E_2} = \frac{V_f}{E_f} + \frac{V_m}{E_m}$$

$$\frac{1}{E_2} = \frac{0.7}{240} + \frac{0.3}{3}$$

$$E_2 = 9.716\ GPa$$

Major Poisson's Ratio:

$$\mu_{12} = \mu_f \times V_f + \mu_m \times V_m$$

$$\mu_{12} = 0.3 \times 0.7 + 0.3 \times 0.3$$

$$\mu_{12} = 0.3$$

Minor Poisson's Ratio:

$$\mu_{21} = \mu_{12} \times \frac{E_2}{E_1}$$

$$\mu_{21} = 0.3 \times \frac{9.716}{168.9}$$

$$\mu_{21} = 0.017$$

Shear Modulus:

$$\frac{1}{G_{12}} = \frac{V_f}{G_f} + \frac{V_m}{G_m}$$

$$G_f = \frac{E_f}{2(1+\mu_f)}$$

$$G_f = \frac{240}{2(1+0.3)}$$

$$G_f = 92.30\ GPa$$

$$G_m = \frac{E_m}{2(1+\mu_m)}$$

$$G_m = \frac{3}{2(1+0.3)}$$

$$G_m = 1.1538\ GPa$$

$$\frac{1}{G_{12}} = \frac{0.7}{92.30} + \frac{0.3}{1.1538}$$

$$G_{12} = 3.737\ GPa$$

Ultimate Longitudinal Strength:

a. Ultimate Failure Strain of Fiber:

$$\varepsilon_{f\,ult} = \frac{\sigma_{f\,ult}}{E_f}$$

Where,

$\sigma_{f\,ult}$ = *Ultimate Tensile Sterngth of Fiber*

$\sigma_{m\,ult}$ = *Ultimate Tensile Sterngth of Matrix*

$\varepsilon_{f\,ult}$ = *Ultimate Failure Strain*

From table,

$$\sigma_{f\,ult} = 4278\ MPa$$

$$\sigma_{m\,ult} = 76\ MPa$$

$$\varepsilon_{f\,ult} = \frac{\sigma_{f\,ult}}{E_f}$$

$$\varepsilon_{f\,ult} = \frac{4278 \times 10^6}{240 \times 10^9}$$

$$\varepsilon_{f\,ult} = 0.017825$$

b. Ultimate Failure Strain of Matrix:

$$\varepsilon_{m\,ult} = \frac{\sigma_{m\,ult}}{E_m}$$

$$\varepsilon_{m\,ult} = \frac{76 \times 10^6}{3 \times 10^9}$$

$$\varepsilon_{m\,ult} = 0.02533$$

c. Ultimate Longitudinal Strength:

$$(\sigma_1^T)_{ult} = (\sigma_f)_{ult} \times V_f + (\varepsilon_f)_{ult} \times E_m \times (1 - V_f)$$

$$(\sigma_1^T)_{ult} = 4278 \times 10^5 \times 7 + 0.017825 \times 3 \times 10^9 \times (1 - 0.7)$$

$$(\sigma_1^T)_{ult} = 3.010\ GPa$$

d. Ultimate Transverse Tensile Strength:

$$(\sigma_2^T)_{ult} = E_2 \times (\varepsilon_2^T)_{ult}$$

$$(\varepsilon_2^T)_{ult} = (\varepsilon_m^T)_{ult} \times \left(1 - V_f^{\frac{1}{3}}\right)$$

$$(\varepsilon_2^T)_{ult} = 0.02533 \times \left(1 - 0.7^{\frac{1}{3}}\right)$$

$$(\varepsilon_2^T)_{ult} = 2.83 \times 10^{-3}$$

$$(\sigma_2^T)_{ult} = 9.716 \times 10^9 \times 2.83 \times 10^{-3}$$

$$(\sigma_2^T)_{ult} = 27.58\ MPa$$

Minimum Fiber Volume Fraction:

$$V_{f\,min} < \frac{(\sigma_m)_{ult} - E_m \times (\varepsilon_f)_{ult}}{(\sigma_f)_{ult} - E_m \times (\varepsilon_f)_{ult} + (\sigma_m)_{ult}}$$

$$V_{f\,min} = \frac{76 \times 10^6 - 3 \times 10^9 \times 0.017825}{4278 \times 10^6 - 3 \times 10^9 \times 0.017825 + 76 \times 10^6}$$

$$V_{f\,min} = \frac{22.52 \times 10^6}{4.3 \times 10^9}$$

$$V_{f\,min} = 0.5238\%$$

Critical Fiber volume Friction:

$$V_{f\,critical} < \frac{(\sigma_m)_{ult} - E_m \times (\varepsilon_f)_{ult}}{(\sigma_f)_{ult} - E_m \times (\varepsilon_f)_{ult}}$$

$$V_{f\,critical} = \frac{76 \times 10^6 - 3 \times 10^9 \times 0.017825}{4278 \times 10^6 - 3 \times 10^9 \times 0.017825}$$

$$V_{f\,critical} = 0.5332\%$$

Shear Strength:

$$\tau_{12\,ult} = G_{12} \times \gamma_{12\,ult}$$

Where, $\gamma_{12\,ult} = \frac{\tau_{12\,m\,ult}}{G_m}$

$$\gamma_{12\,ult} = \frac{56 \times 10^6}{1.1538 \times 10^9}$$

$$\gamma_{12\,ult} = 48.53 \times 10^{-3}$$

The Fiber diameter to fiber spacing ratio is,

$$\frac{d}{s} = \sqrt{\frac{4V_f}{\pi}}$$

$$\frac{d}{s} = \sqrt{\frac{4 \times 0.7}{\pi}}$$

$$\frac{d}{s} = 0.94406$$

Therefore the shear strength is given by,

$$\tau_{12\,ult} = G_{12} \times \left[\frac{d}{s} \times \frac{G_m}{G_f} + \left(1 - \frac{d}{s}\right)\right] \times \gamma_{12\,m\,ult}$$

$$\tau_{12\,ult} = 3.737 \times 10^9 \times [0.0118 + (0.06)] \times 48.53 \times 10^{-3}$$

$$\tau_{12\,ult} = 13.021\ MPa$$

Using the formula of micromechanical analysis of lamina following properties are calculated which is shown in following table.

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Composite Properties	Symbol	Unit	Values
Young's Modulus (Longitudinal Direction)	E_1	GPa	198.9
Young's Modulus (Transverse Direction)	E_2	GPa	9.716
Major Poission's Ratio	μ_{12}	-	0.3
Minor Poission's Ratio	μ_{21}	-	0.017
Shear Modulus	G_{12}	GPa	3.737
Ultimate Longitudinal Strength	$(\sigma_1^T)_{ult}$	MPa	3010
Ultimate Transverse Strength	$(\sigma_2^T)_{ult}$	Mpa	27.58
Ultimate Longitudinal Compressive Strength	$(\sigma_1^C)_{ult}$	Mpa	81.6
Ultimate Transverse Compressive Strength	$(\sigma_2^C)_{ult}$	MPa	21.36
Minimum Fiber Volume Fraction	$V_{f\ min}$	%	0.5238
Critical Fiber Volume Fraction	$V_{f\ cr}$	%	0.5332
Shear Strength	τ	MPa	13.021

Table 6:- Properties of Composite Lamina

VI. ANALYTICAL CALCULATION AND OPTIMIZATION

Our primary objective is to obtain the best layer combination first we started with manual calculations. In carbon fiber the number of layers, orientation of the ply decide the strength of the laminate. There are a large number of combinations that are possible. Manual calculations for every different configuration would be trivial and time consuming. So we wrote a MATLAB code for going through different number of layers (3 onwards) with varying ply orientation. Around 67 lakhs different combinations are checked. Then the stacking sequence satisfying are listed. Depending on the design requirements we have further shortlisted the layers. Lastly FEA analysis is done and the final stacking sequence is selected depending on the MATLAB result and FEA result and manufacturing considerations.

A. Analytical Calculations and Optimization using Matlab software

We developed a MATLAB code referring to the book AUTAR K KAW. The code gives the layers which satisfy the following criteria

- Factor of Safety (Strength Ratio) >2
- Symmetric: configuration is preferred as it eliminates the moment components and analysis is easy

The input to the MATLAB code is the Torque is to be transmitted.

The following output was obtained

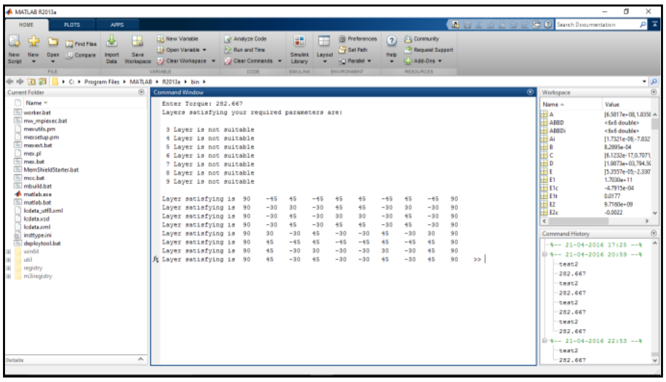

Fig 1:- MATLAB module output window From the output of the MATLAB program following are satisfying layers obtained-

1. 90 -45 45 -45 45 45 -45 45 -45 90 **2.16**
2. 90 -30 30 -30 45 45 -30 30 -30 90 **2.08**
3. 90 -30 45 -30 30 30 -30 45 -30 90 **2.08**
4. 90 -30 45 -30 45 45 -30 45 -30 90 **2.25**
5. 90 30 -30 45 -30 -30 45 -30 30 90 **2.08**
6. 90 45 -45 45 -45 -45 45 -45 45 90 **2.16**
7. 90 45 -30 30 -30 -30 30 -30 45 90 **2.08**
8. 90 45 -30 45 -30 -30 45 -30 45 90 **2.25**

Hence 8 configurations of 10 sequences were obtained from the calculation which satisfy the criteria. We eliminated the stacking sequences with lower strength ratio. The remaining configurations are:

1. 90 -45 45 -45 45 45 -45 45 -45 90 **2.16**
2. 90 -30 45 -30 45 45 -30 45 -30 90 **2.25**
3. 90 45 -30 45 -30 -30 45 -30 45 90 **2.25**

These three layers were analyzed using FEA and their FOS was found out.

1. 90 -45 45 -45 45 45 -45 45 -45 90 **1.70**
2. 90 -30 45 -30 45 45 -30 45 -30 90 **1.55**
3. 90 45 -30 45 -30 -30 45 -30 45 90 **1.66**

From the results the following stacking sequence has the highest factor of safety of 1.70 by FEA and hence it was selected for designing the composite drive shaft.

90 -45 45 -45 45 45 -45 45 -45 90

This stacking sequence is used for final analysis using LAMINATOR software.

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B. Composite Ply Orientation

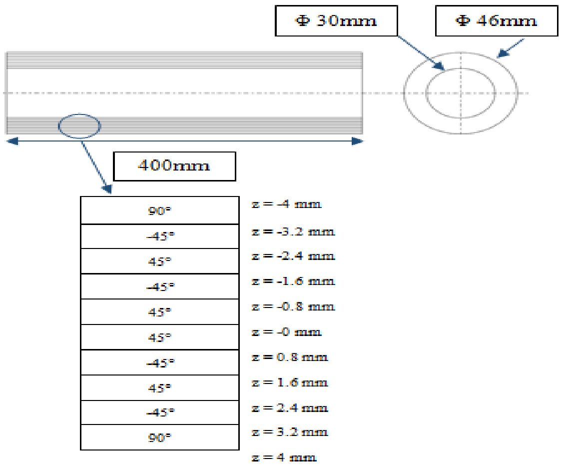

Fig 2:- Arrangement of eight-ply laminates

C. Torsional strength of Shaft

Assuming drive shaft to be a thin, hollow cylinder, an element in the cylinder can be assumed to be flat laminate. The only nonzero load on this element is the shear force, N_{xy} if the average shear stress (τ_{xy}) average the applied torque then,

$$T = (\tau_{xy})_{average} \times \pi(r_o^2 - r_i^2) \times r_m$$

Then

$$N_{xy} = (\tau_{xy})_{average} \times t$$

$$N_{xy} = \frac{T}{2\pi r_m^2}$$

The mean radius, r_m is

$$r_m = r_o - \frac{t}{2}$$

$$= 23 - 4$$

$$r_m = 19$$

$$N_{xy} = \frac{282.667}{2\,\pi\,(0.019)^2}$$

$$N_{xy} = 125.060 \; x \; 10^3 \text{ N}/m$$

The average shear stress is given by

$$N_{xy} = (\tau_{xy})_{average} \times t$$

$$(\tau_{xy})_{average} = \frac{N_{xy}}{t}$$

$$(\tau_{xy})_{average} = \frac{125.060 \times 10^3}{8 \times 10^{-3}}$$

$$(\tau_{xy})_{average} = 15.63 \times 10^6 \; Pa$$

$$(\tau_{xy})_{average} = 15.63 \; Mpa$$

Using LAMINATOR software with stacking sequence $[90,-45,45,-45,45]_s$ and input N_{xy}= 125060 N/m we obtain the ABD matrix and strength ratios of each layer.

D. ABD Matrix

The [A], [B], and [D] matrices are called the extensional, coupling, and bending stiffness matrices, respectively. The extensional stiffness matrix [A] relates the resultant in-plane forces to the in-plane strains, and the bending stiffness matrix [D] relates the resultant bending moments to the plate curvatures. The coupling stiffness matrix [B] couples the force and moment terms to the mid-plane strains and mid-plane curvatures.

Following are the steps followed to find the ABD matrix from LAMINATOR:

1. Input the Laminate elastic properties.
2. Input the Laminate construction: ply thickness, ply orientation, stacking sequence.
3. Calculate the laminate stiffness matrix from the laminate elastic properties.
4. Transform the stiffness matrix to different ply orientation.
5. Calculate laminate stiffness matrices [A], [B], [D].
6. Calculate Laminate strains and stresses.

Once we get the [A] stiffness matrices we can get the values of E_x and E_y. The ABD matrix is obtained from LAMINATOR software for the finalized 10 layer stacking sequence.

E. Ply-by-ply failure analysis of laminate

A laminate will fail under increasing mechanical and thermal loads. The laminate failure, however, may not be catastrophic. It is possible that some layer fails first and that the composite continues to take more loads until all the plies fail. Failed plies may still contribute to the stiffness and strength of laminate.

Following table shows the strength ratio values obtained for each layer in the stacking sequence obtained from LAMINATOR;

Layer	Max Stress	Max Strain	Tsi Hill	Hoffman	Tsai-Wu
1	7.90	7.90	7.90	7.90	7.90
2	2.28	2.25	2.21	2.31	2.25
3	13.54	9.53	13.34	43.17	30.20
4	2.28	2.25	2.21	2.31	2.25
5	13.54	9.53	13.34	43.17	30.20
6	13.54	9.53	13.34	43.17	30.20
7	2.28	2.25	2.21	2.31	2.25
8	13.54	9.53	13.34	43.17	30.20
9	2.28	2.25	2.21	2.31	2.25
10	7.90	7.90	7.90	7.90	7.90
Min	2.28	2.25	2.21	2.31	2.25

Table 7:- Strength ratio values for each layer

From the above table we got minimum Strength Ratio of 2.21 for layer having -45o orientation. This confirms that -45o layers will fail first if the load on that layer increases beyond maximum allowable load, which is given by:

$$\text{Max. allow. load} = Strength\ Ratio\ x\ Applied\ Load$$

$$= 2.21 \times (125.060 \times 10^3)$$

$$= 276.382 \times 10^3 \text{ N}$$

Thus if the load on the -450 layers increase beyond 276.382 kN then layer will fail first.

F. Torsional Buckling Capacity of Shaft

When a hollow shaft is subjected to torsion, at a certain amount of torsional load instability occurs. This is called the torsional buckling load. An orthotropic thin hollow cylinder will buckle torsionally, if the applied torque is greater than the critical torsional buckling load given by:

$$Tc = (2\pi r_m^2\, t)(0.272)\left(E_x \times E_y^3\right)^{\frac{1}{4}} \times \left(\frac{t}{r_m}\right)^{\frac{3}{2}}$$

From the Laminator Software and by using Macro mechanical and micromechanical analysis of lamina, we get the longitudinal young's moduli Ex and the transverse young's moduli E_y of the [90 -45 45 -45 45 45 -45 45 -45 90]$_s$ carbon fiber/epoxy laminate based on calculation and from laminator software,

$$Ex = \frac{1}{h \times A_{11}^*} = \frac{1}{0.8 \times 4.829e-009} = 25.88\ GPa$$

$$Ey = \frac{1}{h \times A_{22}^*} = \frac{1}{0.8 \times 2.738e-009} = 45.65\ GPa$$

Because lamina thickness is 0.8mm, the total thickness of eight ply is

$$t = 10 \times 0.8$$

$$t = 8mm$$

$$Tc = (2\pi r_m^2\, t)(0.272)\left(E_x \times E_y^3\right)^{\frac{1}{4}} \times \left(\frac{t}{r_m}\right)^{\frac{3}{2}}$$

$$Tc = 53416.1\ Nm$$

This value is greater than the applied torque of 283.67 Nm, thus the composite shaft is safe in buckling.

G. Torsional Stiffness

To find the torsional stiffness and angular deflection of drive shafts following steps are followed;

From LAMINATOR we got, $G_{xy} = 9.576\ GPa$

L = 0.4 m

$$J = \left(\frac{\pi}{32}\right) \times (d_o^4 - d_i^4)$$

$$J = \left(\frac{\pi}{32}\right) \times (0.046^4 - 0.030^4)$$

$$J = 3.6 \times 10^{-7}\ m^4$$

Hence, torsional stiffness is

$$K_t = \frac{GJ}{L} = \frac{9.576 \times 10^9 \times 3.6 \times 10^{-7}}{0.4}$$

$$K_t = 8615.26\ Nm/rad$$

Thus torsional deflection is,

$$\theta = \left(\frac{T}{K_t}\right) = \left(\frac{282.667}{8615.26}\right) = 0.328\ rad = 1.88\ Deg./0.4m$$

$$\theta = \frac{1.88}{0.4} = 4.7\ Deg/m$$

H. Natural Frequency

Find out the minimum Natural Frequency of the drive shaft, given by

$$f_{nt} = \frac{\pi}{2}\sqrt{\frac{EI_x}{mL^4}}$$

$$f_{nt} = \frac{\pi}{2}\frac{\sqrt{(25.88 \times 10^9)(1.8 \times 10^{-7})}}{\sqrt{1.475 \times 0.4^4}}$$

$$f_{nt} = 551Hz$$

Since the minimum Natural frequency required is 50 Hz, this requirement is also met by our laminate.

The critical speed of shaft is given by,

$$N_{crt} = 60 \times f_{nt}$$

$$N_{crt} = 60 \times 551$$

$$N_{crt} = 33060\ rpm$$

I. Mass Saving

- Mass of steel drive shaft = 1.15Kg
- Mass of Composite drive shaft calculated:

$$m = \pi\,(r_o^2 - r_i^2)L\rho$$

$$m = \pi\,(0.023^2 - 0.015^2)(0.4)(1548)$$

$$m = 0.590\ Kg$$

- Percentage of mass saving over steel:

$$= \frac{1.15 - 0.590}{1.15}\ x\ 100$$

$$= 48.69\ \%$$

J. Result summary of designed Composite drive shaft

To find the torsional stiffness and angular deflection of drive shafts following steps are followed;

Sr. No.	Parameters	Composite Shaft (Designed)
1	Outer Diameter	46 mm
2	Thickness	8 mm
3	Applied Torque (T)	282.667 Nm
4	Torsional Buckling (Tb)	53416.1 Nm
5	Natural Frequency (f_{nb})	551 Hz
6	Critical Speed (N_{cr})	33060 rpm
7	Torsional Stiffness (K_t)	8615.63 Nm/rad
8	Carbon fiber: epoxy (% vol)	70 : 30
9	Mass (m)	0.590 Kg
10	Percentage of mass saving	48.69 %

Table 8:- Design of Composite Drive Shaft

VII. ANALYSIS

A. Finite element analysis

FEA is a computational tool for performing engineering analysis. It includes the use of mesh generation techniques for dividing a complex problem into small elements, as well as the use of software program coded with FEM algorithm. In applying FEA, the complex problem is usually a physical system with the underlying physics such as the Euler-Bernoulli beam equation, the heat equation, or the Navier-Stokes equations expressed in either PDE or integral equations, while the divided small elements of the complex problem represent different areas in the physical system.

B. Analysis of Composites

The analysis of composite shaft is done using SolidWorks Simulation. The aim is to optimize orientation of the composite ply layup to ensure product quality, performance, and factor of safety (FOS).

The failure criterion for composite materials is very different than for metals. Composite materials do not yield; rather, the fibers delaminate and fracture. SolidWorks Simulation reports the Factor of Safety (FOS) against failure according to the Tsai-Wu, Tsai-Hill and Maximum shear stress failure indexes.

SolidWorks Simulation uses finite element analysis (FEA) methods to discretize composite components into shell elements and uses stress analysis to determine the response of parts and assemblies due to the effect of:

- Forces
- Pressures
- Accelerations
- Temperatures
- Contact between components
- Fiber Delamination

C. Shaft Modelling in FEA software

In order to perform the analysis in FEA software, we need to create the model of shaft as per the analytical design. Then we need to provide all the details of carbon fiber, resin used, in order to find the composite properties. The next step is to input the different layers in stacking of carbon fiber shaft. Further the meshing of shaft is done and required constraints are applied. After constraints the torsional loading is done on shaft and analysis is performed. Based on the analysis results stacking sequence with lower factor of safety are eliminated and one with maximum factor of safety is finally selected.

D. Results obtained from FEA

The FEA results obtained for final shortlisted stacking sequence i.e. $[90\ \ -45\ \ 45\ \ -45\ \ 45]_s$ is as follows;

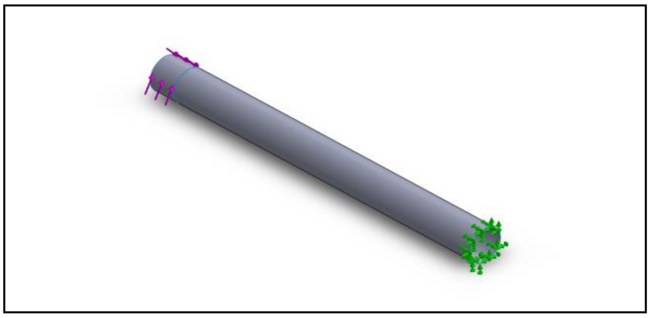

Fig 3:- Modelling of Composite drive shafts with constraints and loading

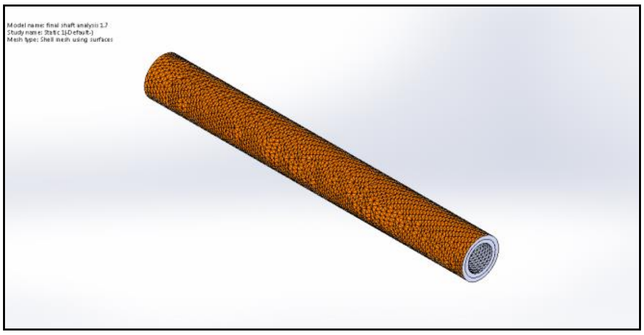

Fig 4:- Meshed model of drive shaft

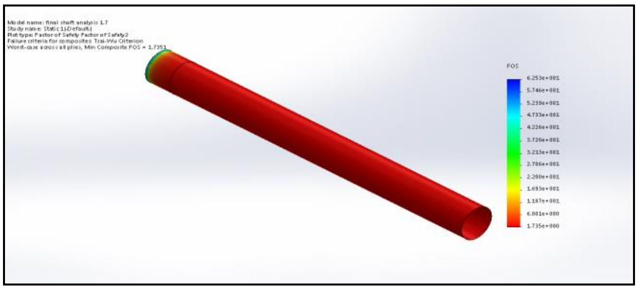

Fig 5:- FOS by Tsai-Wu Theory – [1.702]

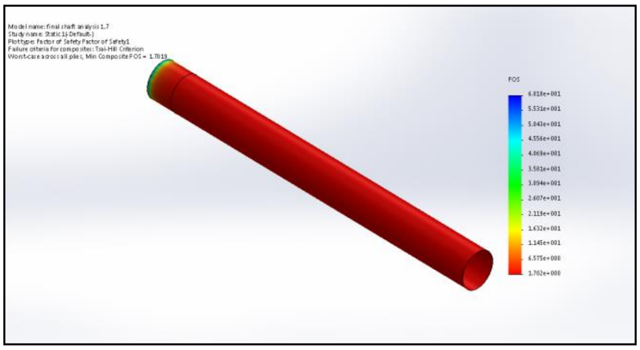

Fig 6:- FOS by Tsi-Hill Theory – [1.735]

Fig 7:- FOS by Maximum Stress Theory – [1.740]

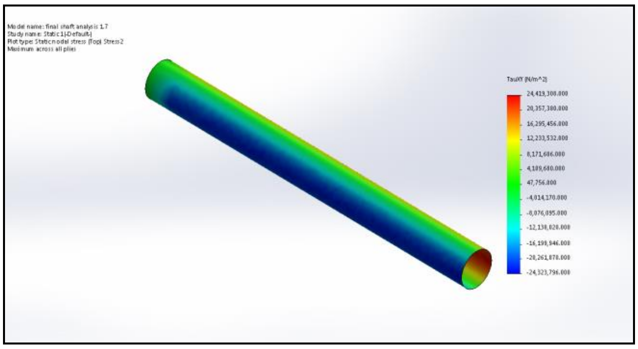

Fig 8:- Shear stress generated – [24.419 MPa]

E. Conclusion of FEA

The Shear stress generated in shaft is 24.419 MPa.
T_{si} Wu = 1.735, T_{si} Hill = 1.702, Max Stress = 1.740

VIII. MANUFACTURING

A. Filament Winding Process

In filament winding method, fiber strands are unwind and passed continuously to the resin tank. In resin tank, fiber strand are impregnated completely with the resin. Now, these resin impregnated strands are passed onto a rotating mandrel.

These strands are wound around the mandrel in a controlled manner and in a specific fiber orientation.

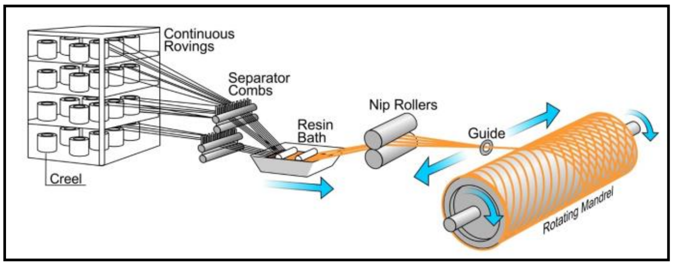

Fig 9:- Filament Winding Process

B. Manufacturing

The shaft is manufactured using fully automated CNC Delta filament winding machine at the facility of Carbon Light Pvt. Ltd., Ghaziabad, U.P., India. The machine uses four spools of Panex 35 simultaneously and orients the spool as programmed by the programmer.

IX. TORSION TEST

A torsion test can be conducted on most of the materials to determine the torsional properties of the material like modulus of elasticity in shear, yield shear strength, ultimate shear strength, modulus of rupture in shear, ductility, etc. Torsion testing is performed since many products and components are subjected to torsional forces during their operation. Products such as biomedical catheter tubing, switches, fasteners, and automotive steering columns are just a few devices subject to such torsional stresses. By testing these products in torsion, manufacturers are able to simulate real life service conditions, check product quality, verify designs, and ensure proper manufacturing techniques.

A. Procedure

For the torque testing of the shaft, end attachments are required to be fixed on the shaft for its proper mounting on the torque testing machine. Steel ends are made which are press fitted inside the shaft. Aeronautical Grade Adhesive is applied at the end attachments before they are inserted in the shaft. The adhesive makes sure that the end attachments are firmly attached with the shaft transmitting the required torque. The end attachments are then bolted to the torque testing machine. SYSCON Torsional testing machine is used for the testing purpose. The machine can apply a maximum load of 10,000 Nm. The testing is done at the facility of Carbon Light Pvt. Ltd. Ghaziabad, U.P. India.

Fig 10:- Torque Testing of Composite Shaft

B. Results of Torsion Test

Following results were obtained from the testing:

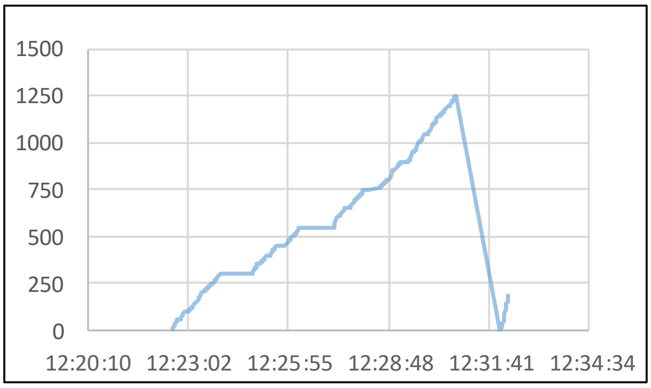

Fig 11:- Torque Vs Time

From the above graph it can be seen that after 1250 Nm the shaft stops transmitting the torque.

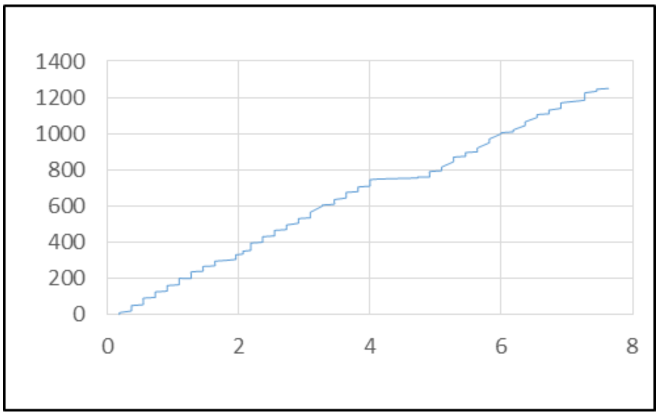

Fig 12:- Torque vs Angle of twist graph

The deflection increases proportionally with the torque up to 720 Nm. At 720 Nm there is very less change in torque but a large change in deflection which is due to loss of elastic behavior in angle of twist.

C. Torsional Stiffness Calculation

Torsional stiffness in terms of torque is given by,

$$K_t = T/\theta$$

From the graph we found out value of K_t as follows;

Taking two points from graph (1.636, 280) & (2.364, 400)

$$K_t = \frac{T_2 - T_1}{\theta_2 - \theta_1}$$

$$= \frac{400 - 280}{2.364 - 1.636}$$

$$= 164.84 \frac{Nm}{deg}$$

$$K_t = 9444.63\ Nm/rad$$

Hence torsional stiffness obtained experimentally is 9444.63 Nm/rad.

D. Angular Deflection

$$\theta = \left(\frac{T}{K_t}\right) = \left(\frac{282.667}{9444.3}\right) = 0.0299\ rad = 1.71\ Deg./0.4m$$

$$\theta = \frac{1.71}{0.4} = 4.2\ Deg/m$$

E. Modulus of Rigidity

$$G = \frac{LK_t}{J} = \frac{9444.3 \times 0.4}{3.6 \times 10^{-7}}$$

$$G = 10.49 \times 10^9 \ Pa$$

F. Comparison Table

Sr. No.	Parameters	Composite Shaft (Manufactured)
1	Outer Diameter	46 mm
2	Thickness	8 mm
3	Applied Torque (T)	282.667 Nm
4	Torsional Stiffness (K_t)	9444.3 Nm/rad
5	Carbon fiber: epoxy (%)	65 : 35
6	Angular Deflection (θ)	4.2 deg/m
7	Mass (m)	0.650 kg
8	Percentage of mass saving	44.26 %

Table 9:- Properties of Manufactured Composite drive shaft

X. CONCLUSION

In this project firstly we designed the composite drive shaft using Classical Lamination theory. We optimized the stacking sequence with the help of MATLAB and FEA software Solidworks. After the manufacturing, torsional testing was conducted. Also we found out the actual volumetric ratio of fiber and resin. From the above study following conclusions were made:

- The carbon fiber drive shaft has been designed to replace the steel drive shaft of the SAE Baja ATV. The shaft has been optimally designed with the objective of minimizing the weight of the shaft which was subjected to the constraints such as torque transmission and outer diameter.
- The weight saving of the manufactured carbon fiber drive shaft is 44.26 % as compared to steel drive shaft. There is a deviation of 4.43% between the designed and the actual weight saving of the composite drive shafts.
- The calculated buckling torque at which the shaft will fail is very high as compared to the actual buckling torque acting on the shaft.
- The deflection of the manufactured shaft is 4.2deg/m which is less than the deflection obtained by analytical calculation which was 4.7deg/m.

Parameter	Steel shaft	Composite Shaft (Designed)	Composite Shaft (Manufactured)
Outer Diameter	37 mm	46 mm	46 mm
Thickness	2.5 mm	8 mm	8 mm
Applied Torque (T)	282.667 Nm	282.667 Nm	282.667 Nm
Torsional Buckling (Tb)	33788.149 Nm	53416.1 Nm	-
Natural Frequency (f_{nb})	606.583 Hz	551 Hz	-
Critical Speed (N_{cr})	36352.3 rpm	33060 rpm	-
Torsional Stiffness (K_t)	6174.46 Nm/rad	8615.63 Nm/rad	9444.3 Nm/rad
Carbon fiber: epoxy (%)	-	70 : 30	65 : 35
Angular Deflection (θ)	2.32 deg/m	4.7 deg/m	4.2 deg/m
Mass (m)	1.15 Kg	0.590 Kg	0.650 kg
Percentage of mass saving	-	48.69 %	44.26 %

Table 10:- Properties of Manufactured Composite drive shaft

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